COMPARATIVE MEASUREMENTS BEHIND A COMPRESSOR ROTOR WITH A SPACE-BOUND AND COROTATING CYLINDRICAL PROBES

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16. Abstract

The total pressure and the flow angle at various distances from the axis were studied for four rotor speeds. Data obtained with a cylindrical probe which rotated with the rotor are compared with values indicated by the same probe kept in a fixed position. No significant differences concerning the mean total pressure, or the flow angle at great distances from the axis could be observed for both probe positions. There were, however, differences in the flow angle amounting to as much as 2 deg in locations near the axis.

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COMPARATIVE MEASUREMENTS BEHIND A COMPRESSOR ROTOR
WITH A SPACE-BOUND AND COROTATING CYLINDRICAL PROBES

A. Klein[1]

1. Introduction

The problem of forming averages for time-variable pressure has great significance for research in testing the flow processes on turbine blades. Both the pressure and flow angle are mainly determined by use of fixed probes which, as a result of the backwash of the blades, receive "waves" periodically over time. Little is known how much the values so measured agree with the true average temporal values. In order to avoid this source of error, various researchers have employed wake probes, which revolve at the same rpm as the rotor. They have thus measured the relative flow over a blade section, and then have determined a mean value. Such measurements were conducted, e.g. by E. Muhlemann [1], G. Muesmann [2], W. Dettmering [3] and also W. M. Schulze, et al [4] with conventional probes. H. Ufer [5] used a rotating hot wire probe. Studies of this kind are very expensive, both in respect to measuring technology required and in evaluating the results. It therefore is important to clarify beforehand the possible errors in measuring with fixed probes for the conditions which occur in compressors. /25*

A number of theoretical studies exist in this problem area. However, a number of simplifying assumptions have to be made, both for the form of pressure changes over time and for the shape of the probe and measuring leads. These do not allow a direct transfer of the findings to the given conditions in compressors. M. V. Nesbitt [6] has shown that for an incompressible fluid under pressure, whose periodic time curves are square, saw-toothed or sinusoidal, the

[1]This investigation was undertaken in the course of the author's work and the Institute for Aerodynamics of the German Research Institute for Aeronautics and Astronautics, at Braunschweig.
*Numbers in the margin indicate pagination in the foreign text.

true temporal mean value can be indicated. For a compressible fluid M. V. Nesbitt has deduced relations between the indicated and the real mean pressure. Admittedly, these considerations are valid only when the mass of the measuring apparatus for the most part lies behind the probe bore, and has a much greater cross-section. For measurements in compressors, however, this principle can not be realized because of space considerations.

R. C. Johnson [7] has investigated total pressure probes with very long bores. If they are long enough that end effects play no role, at least for the laminar flow of an incompressible fluid, the throughflow is a linear function of the applied pressure. The probe then shows the actual temporal average. Deviations become greater the relatively shorter the probe bore. With compressible fluids, in contrast, even with long bores the average temporal value is not obtained since the throughflow also is a function of density, which is dependent upon the applied pressure. For laminar throughflow (at) the probe bore R. C. Johnson [7] has derived relations, with the help of which one can determine the true temporal mean value of a total pressure which fluctuates periodically between two constant values from the value indicated by the probe as a function of the form of the pressure perturbation and the shape of the probe. These relations correlate well with measured results; they are valid, however, only for pressure functions which have a square periodic temporal curve. Still, these investigations have shown that, for an incompressible fluid, only small errors in the indicated total pressure are to be expected, if one chooses a large enough ratio of length to diameter for the probe bore and designs it so that both the in- and outflows are laminar. The long Pitot tubes used by R. C. Johnson of course are not suitable, because of their space requirements, for measurements between the blades of compressors.

Besides the shape of the probe geometry, the dimensions of the leads to the measuring instrument also have great significance for the error of the pressure readings. In order to keep it small or avoid it entirely, the lead system must include a throttle which suppresses pressure fluctuations. The cross-section behind it must be as large as possible, so that the amplitudes of the fluctuations which are carried to the manometer remain small. Studies on a suitable design for the throttle position have been made e.g. by H. G. Heinrich [8]. It must be directly at the place of measurement and hence be identical with the measuring bore of the probe. If this is not the case, then resonance could occur in the lead section between the measuring point and the throttle, and this falsifies the reading. The same danger exists if the pressure fluctuations in the lead system behind the throttle are too large, i.e. if it is too small in cross-section. L. J. Kastner e.g. [9] has described measurement error at the manometer as result of resonances in the measuring leads. Although this error can be calculated in general, according to a procedure of I. Taback [10], it is better to design initially the lead system so that resonances are avoided.

For total pressure measurements behind the rotors of compressors, these considerations signify that, in measuring with fixed probes, the following criteria have to be applied at the start: One should choose a ratio of length to diameter of the probe bore as large as possible and the shape so that the /26
throughflow in both directions is laminar, as well as providing the greatest possible cross-section behind this bore. In the measuring leads no additional throttle points may exist. Further, the wavelength of pressure perturbations applied to the probe must be large compared with the length of its bore, and the frequency of the pressure fluctuations must be far above the critical value of the system which consists of the probe and measurement leads, so that resonances are completely avoided.

Falsification of the angle reading, which may occur in measurements with a stationary probe behind a rotor, has not been researched theoretically before. On the other hand, W. M. Schulze et al [4] sought to clarify this problem by experiments with a compressor rotor. They obtained a maximum error of about 1°, which appeared in measuring sections near the hub. At greater distances from the hub, however, these investigators found no error. To be sure, their results cannot be regarded as generally valid since the important parameter of the rotor rpm, and thus the frequency of pressure perturbation, was not varied.

Similar comparative measurements for total pressure have not been made known. The following study is a contribution to the problem of total pressure and angle measurements in compressors. Flow magnitudes have been determined behind a compressor rotor, both with a stationary and with a rotating probe at four different revolution speeds at several shaft distances, and the results were compared with one another. A cylindrical probe served as measuring sensor. This is especially well suited to measuring in compressors because of its small axial extension. The tests were limited to mach numbers at which the fluid can be considered incompressible.

2. The Test Apparatus

Figure 1 shows a schematic to scale of the experimental device. The rotor a was mounted propeller-wise and attached to a section of hub which protruded a ways upstream. Its diameter at the hub was 244 mm and at the housing 444 mm; the hub ratio thus was 0.55. At an axial distance of 30 mm behind the rotor, corresponding to 0.6 profile depth in the direction of flow, the co-rotating probe b was installed. With it, it was possible to simultaneously measure the total pressure p_g and the flow angle β, which the velocity vector forms in the peripheral direction in the relative system. The pressures recorded on the bores of the probe are carried separately via plastic tubes to three small tubes set in the hollow shaft, and then on into the chamber of the pressure conversion device c. There the pressures are converted from the rotating to the fixed system. The leads d connect the chambers with the manometers. A direct-current motor e of 7.5 kW capacity, which is found downstream on the solid part of the hub, drives the rotor. Its rotation speed can be varied continuously. It can be controlled while running with help of the tachometer f. The blower g draws air from the environment via an inlet jet through the experimental device. The throughput necessary for the rpm in each case is regulated with the aid of the secondary vent h, which corresponds to the pressure readings from three calibrated bulkhead venturii i, which are distributed 60 mm in front of the rotor equally around the periphery of the housing.

Figure 2 shows details of the experimental device. The probe holder d is connected firmly to rotor c and rotating hub a with the aid of the two screws g and h, so the rotor may be adjusted peripherally, relative to the probe [sic]. In this way the measurements may be made over an entire blade section in optionally fine gradations. The probe may be shifted radially by loosening a clamp. Imbalance is avoided by counterweight i. Before the beginning of the tests all rotating parts, together with the probe, were balanced accurately. Thereby, the positions of the counterweights on the probe are determined for any distance of the probe head from the hub at which the flow is to be measured.

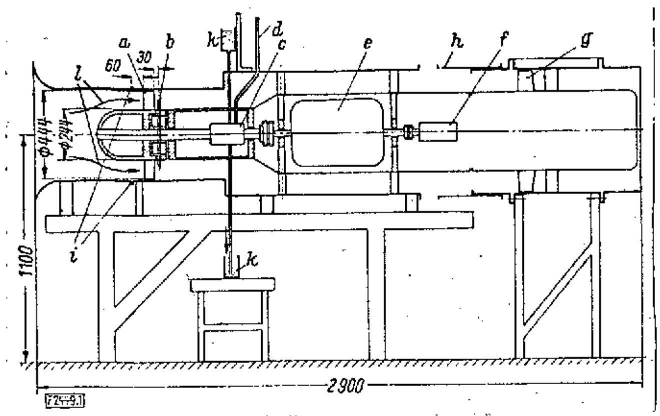


Figure 1. Experimental Device. Dimensions in mm; a rotor, b co-rotating probe, c pressure converting device, d pressure leads to the manometers, e direct current motor, f tachometer, g blower, h secondary vent, i bulkhead venturii for throughput measurement, k oil reservoirs, l flowlines in the scoop.

The angular and peripheral position of the probe relative to the rotor may be changed only at rest. However, the probe may be moved about its long axis and thus set in the direction of flow while rotating. For this there is an adjusting device, partly visible in Figure 2. An external hand crank attached to the experimental device affects, via bevel gears and a spindle n on the fixed outer race of the axially movable hub o on the shaft, the center race of the hub being joined firmly to the shaft. Between these two hub races are ball bearings. On the rotating race is a stud m, which slides in a shaft groove and is guided through under the main bearing l. It is connected to another stud b, part of which holds the probe firmly. If the hand crank is turned, the hub moves axially on the shaft. The stud attached to /27
it transfers the rotary motion then to the probe. This device is constructed so that the probe is turned exactly 1° about its long axis with a full revolution of the crank. The total swing range is 40°, enough for the measuring program. The position of the handcrank is transmitted to a counter, the reading of which permits direct conclusions about the angle of flow. The adjusting mechanism of the probe is similar to that used by G. Muesmann [2]; his test apparatus was not counterbalanced, and therefore permitted only relatively low rpm.

The pressure converting device (p in Figure 2) inside of which the transition from rotation to a fixed system occurs, is detailed in Figure 3. It consists of eight o-rings a mounted in-line on the shaft, between which are ring-shaped chambers. Into three of them pass one each of the small tubes b which pass through the hollow shaft and are connected to the measuring leads of the probe. These chambers are connected to the indicating instruments by means of nipples c and tubes. On each side of each pressure chamber is another chamber through which oil flows for added sealing, lubrication, and above all for cooling (cf. Figure 1). This cooling proved necessary, because at high rpm considerable heat occurred through friction of the seals on the shaft. On the underside of the pressure chambers nipples d also were installed, over which were placed sections of transparent tubing closed at the bottom. These allow checking whether there are oil leaks into the chambers, which would falsify the pressure measurements. The outer part of the pressure converter is connected with the housing by means of the torsion braces e.

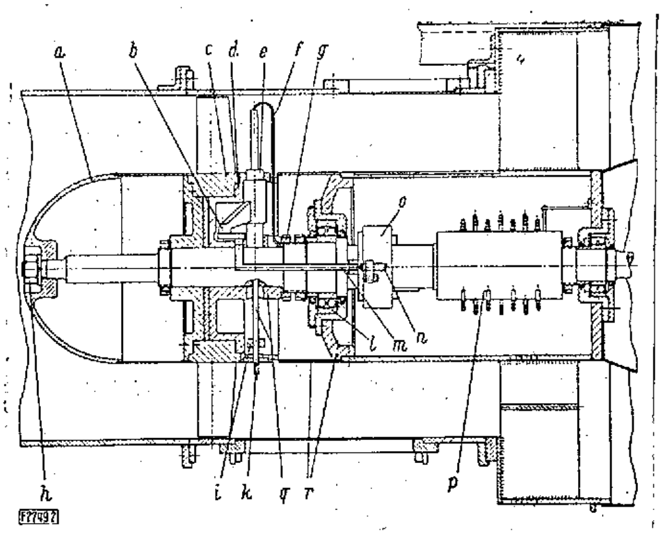


Figure 2. Details of the Experimental Device. a rotating hub, b probe rotation device, c rotor, d probe holder, e clamp arrangement for radial probe adjustment, f pressure measuring leads, g and h female screws to adjust the probe relative to the rotor in the peripheral direction, i counterweight, k co-rotating probe, l main bearing, m lever, n spindle, o axially displaceable hub, p pressure converting device, q rotating part, r fixed part.

Originally it was planned that the o-rings would be symmetrical to the respective chambers, i.e. so that the upturned edge of the rubber sleeve always would be face toward the respective chamber. This produced a better sealing effect than that shown in Figure 3, but experience showed that it is very difficult to push the o-rings over the shaft in reversed position, without damaging the sealing ridges. In the arrangement shown in Figure 3 the ring installation was made from the left. Injuries to the ridges then could only be avoided if the rubber sleeves always pointed toward the left. The pressure in all cases remained low enough so that the o-rings sealed perfectly, nevertheless.

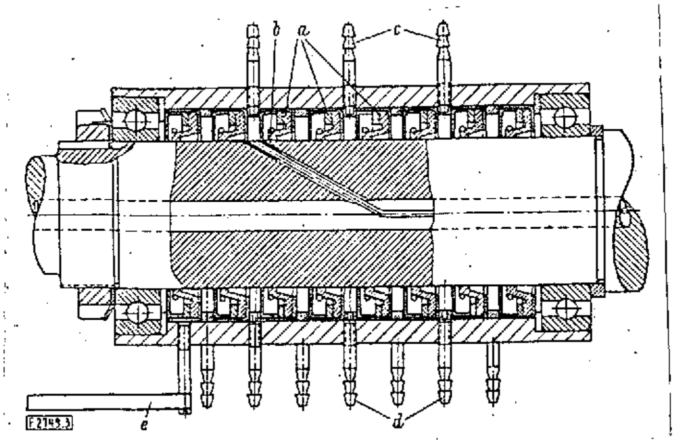


Figure 3. Pressure Converting Device. a o-rings, b small pressure measuring tubes connected with the probe, c tubing nipples, d nipples for attaching closed-end plastic tubes for oil-leak detection, e housing support.

For runs with the fixed probe, a part of the housing of the experimental device replaced by two rings. Between these the wake measuring equipment, in which the cylindrical probe was inserted, was placed. Therefore, the probe could be radially shifted and rotated about its long axis, and it was also possible to read off the respective angle. The measurement plane distance was the same as with the rotating probe.

The compressor rotor used for the studies had 20 blades, which were so twisted that the degree of reaction remained constant over the blade length. The blade angle measured between the profile chord and the peripheral direction was 103.8° at the hub and 129.6° at the housing, the blade depth l = 53.6 mm and the pitch ratio t_N/l = 0.713 at the hub and t_A/l = 1.301 externally on the housing (with t_N and t_A as the pitch at the hub or externally). The profile shapes resemble those of the older NACA-system[2] with a maximum relative density of 10%.

3. Probes Employed

3.1 Cylindrical Probe

A cylindrical probe was chosen for comparative measurements as this type of probe is very often used because of its small axial space requirements in

[2]Such profiles of the older system of the National Advisory Committee for Aeronautics indeed are not suitable for producing high efficiency; but this is not significant for the aim of this study.

studies of compressors. It is also especially suited for experiments in which the probe rotates, since the centrifugal force cannot distort its head. E. Muhlemann [1], e.g., in his tests with a rotating Pitot tube, had to determine the distortion of the probe head with an expensive optical procedure to obtain an exact distance from the axis of rotation. Cylindrical probes also guarantee angular measurements of high precision, which at not very large radial angles are in fact largely independent of the angles. A wedge probe, which also has a rather small axial space requirement and is also used in experiments with compressors, on the other hand often manifests, with small radial angles, a large interdependence of the angular measurement, because uncontrollable sintering processes occur at the edges of the probe.

Furthermore, with a cylindrical probe, the requirement for a relatively long bore to measure the overall pressure can be most readily linked with mechanical resistance to centrifugal forces. This type of probe also has the advantage that its behavior is very precisely known. W. Wuest [11] has summarized the large number of investigations on cylindrical probes (see additional references there). The disadvantage of this type of probe lies solely in the fact that its measuring bore must be a definite distance from the hemispherical head. Thus, measurements cannot be taken too near the walls, but this plays no great role here.

Figures 4a through 4d show the cylindrical probe used. It has one bore for measuring total pressure and two, offset at ±45°, for angular measurements. Each bore is 0.3 mm in diameter. The probe has an outside diameter of 3.5 mm and a 6 mm shaft of circular cross-section. The head is rounded off by a hemisphere. All dimensions lie within the framework of recommendations given by W. Wuest [11]. The provisions for exact measurements were specified therein.

The ratio of length to diameter of the total pressure bore was 5.2:1, which is a relatively large value. The flow through this bore remains laminar for all /28
velocities appearing during the measurements. It is connected to a conduit of 0.8 mm diameter, whose cross-section thus was seven times as large as that of the measuring bore [sic]. After some 30 mm the conduit expands to a 1.3 mm diameter, and finally to 1.5 mm. The dimensions mentioned also coincide with the recommendations of R. C. Johnson [7] within the scope of the possibilities

stated, so that it is possible to plan on only small deviations of the total pressure, measured with the fixed probe, from the mean temporal average, so long as the fluid may be regarded as incompressible. The conduit for angular measurement had the same dimensions. In this, of course, the ratio of length to diameter of the measuring bores is only 1.7:1.

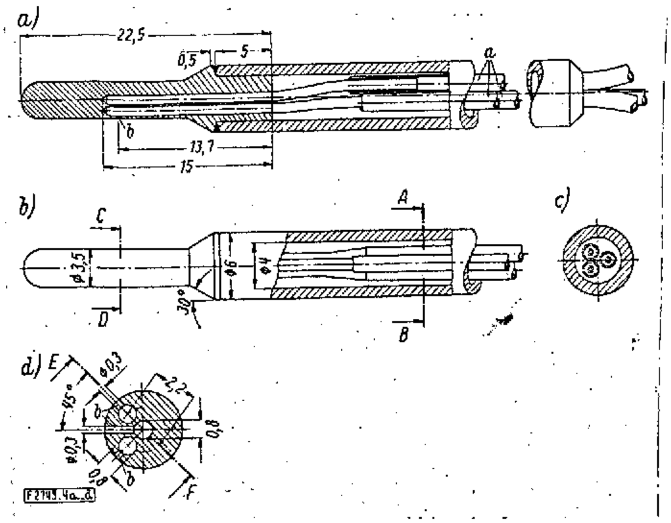
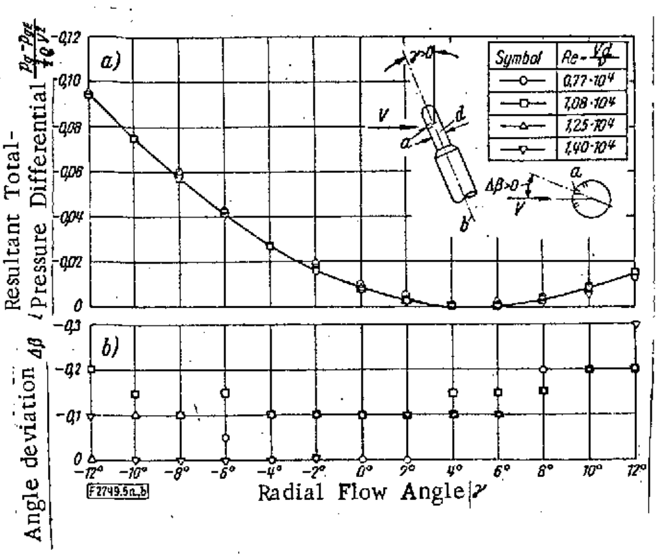


Figure 4. Cylindrical Probe Used. Dimensions in mm; a three 280 mm long tubelets of 1.5 mm diameter with 0.1 mm wall thickness, b measuring bores; a) section E-F, b) rear view of the probe with partial longitudinal section, c) section A-B, d) section C-D at double scale.

The probe was tested for the influence of radial flow angles γ at four Reynolds numbers formed with its diameter, in the range $-12° \leqq \gamma \leqq 12°$. Larger radial angles cannot be expected in the flow fields to be measured. The corresponding standard curves for total pressure and angular reading can be seen in Figures 5a and 5b. Although the radial angle γ has a considerable influence on the total pressure measured, the indicated angle in the range studied causes an error of only $\Delta\beta = \pm 0.1°$ of the relative flow angle β.

3.2 Other Probes

Cylindrical probes are unsuitable for the measurement of static pressure, in case the flow field, as in the case discussed, exhibits a velocity gradient in the direction of its axis [11]. Therefore, a conventional hook-shaped static pressure probe, which also has false readings resulting from radial flow components estimated, is used. A hemispherical probe with two bores, one over the other, serves to measure the radial angle. In this arrangement the angle may be determined from the pressure differential read between the two bores, using of a standard curve. Figures 6a and 6b show the two probes. As is evident they are so shaped that measurements may be made with these probes the same distance behind the rotor, as with the cylindrical probe. The axial position of the wake-measuring device may be ȧdjusted apporpriately in the experimental model housing.

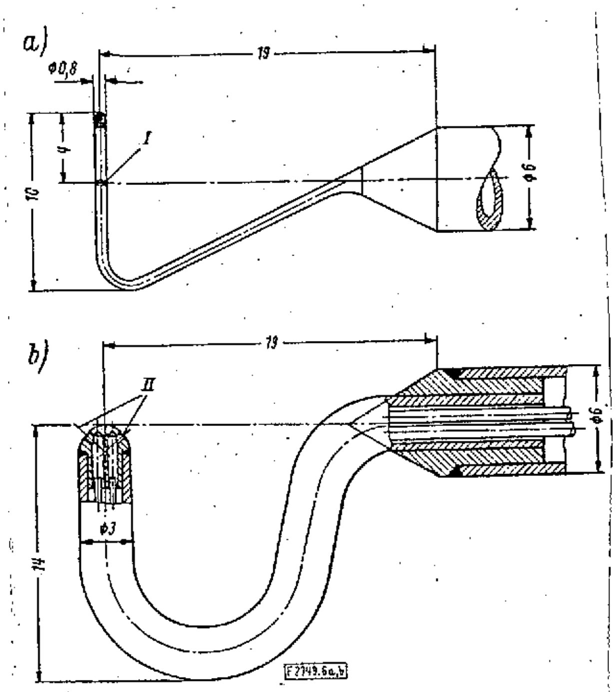


Figures 5a and 5b. Standard Curves of the Cylindrical Probe. P_g and P_{gE} indicated total pressure of the cylindrical probe and standard probe; $qV^2/2$ the kinetic pressure formed with density q and flow velocity V of the fluid; $\Delta\beta$ and γ the angle between the flow direction and the axis a of the total pressure bore and between the norms at V and the axis b of the probe; Re the Reynolds number formed with flowrate V, probe diameter d and the kinematic viscosity ν of the fluid; a) standard curve for total pressure reading, b) standard curve of the angle reading.

4. Making and Evaluating the Measurements

With the rotating cylindrical probe, wake measurements were made over a blade section at rpm's of n = 1000, 1350, 1700 and 2050 rpm at five radial distances as per r/r_A = 0.590, 0.703, 0.815, 0.883 and 0.950 (with r as the axial distance and r_A the radius at the blade tips). The standard rpm was controlled constantly with the jet tachometer and held constant to $\Delta n = \pm 10$ rpm. The throughput was so set that the value obtained was the same for all tests.

A Prandtl manometer indicated the total pressure registered by the probe. The relative flow angle β_2 behind the rotor was obtained with help of the null method. Measurement precision for $\Delta\beta_2 = \pm 0.1°$, and $\Delta\beta_2$ could be reproduced within ±0.3°. To determine the level from which the angle was determined, the probe was oriented in the stream of a small wind tunnel to be exactly horizontal and then a small precision water-level was mounted on it. With its help, the probe could be so installed in the experimental model, that the bore axis formed an exact 90° angle with the peripheral plane. Before a series of tests was begun, the water-level was removed. If the probe was /29 radially displaced to measure a new section, the above process was repeated. During the experimental program the seal of the pressure converter was checked constantly, both with the motionless and the rotating probe. There were no malfunctions.



Figures 6a and 6b. Other Probes Used. Dimensions in mm; a) probe for measuring static pressure with four bores at I; b) probe to measure radial angles with two bores at II.

After conclusion of these measurements, the total pressure and angle were determined, at the same rpm and throughput in the absolute system with the same cylindrical probe in fixed arrangement at seven axial intervals. Finally, with the probes shown in Figures 6a and 6b the static pressure and radial flow angle were measured.

The Reynolds number Re_1 formed with the relative velocity $\underline{V}$ in front of the rotor at profile depth l lay between $Re_1 = 0.8 \cdot 10^5$ at the smallest axial distance with $\underline{n}$ = 1000 rpm and $Re_1 = 2.2 \cdot 10^5$ at the largest axial distance with $\underline{n}$ = 2050 rpm. The velocities were low enough so that the fluid could be regarded as incompressible.

The differential of the total pressure P^*_{g2man} to the ambient pressure P^*_{at}, which the manometer read when measuring with the rotating probe, varied from the differential of the true total pressure P^*_{g2} to P^*_{at} by the kinetic pressure of the peripheral velocity of a part of the air volume[3] inclosed in the probe.

[3]Here the index 2 on the total pressure P_g refers to the value behind the rotor, the index 'man' to the manometer reading, and the asterisk to the co-rotating measuring system.

Thus, recalculation was necessary according to the equation

$$p_{g2}^* - p_{at}^* = p_{g2Man}^* - p_{at}^* + \frac{\varrho^*}{2} \omega^2 (r^2 - r_0^2) \qquad (1)$$

There q* indicates density of the fluid, ω the angular velocity of the probe, r the radial axial distance of the probe bore and r_0 the axial distance of the measurement leads in the chamber of the pressure converted. Recalculation, according to Equation (1) assumes adiabatic lamination of the air column inclosed in the probe. Research by K. Leist and W. Dettmering [12] has shown that this assumption is valid at the relatively low velocities existing here.

The absolute and relative total pressures and the static pressure are corrected with the corresponding standard curves of the probe as regards errors due to radial flow components. Also, the relative total pressure in the resting state, as determined by the ambient pressure and temperature, is reduced to that prevailing during the test with the stationary probe. The values measured in the relative and absolute systems may then be compared with each other only after reduction to a general system. W. Dettmering [3] has stated the relationship for this. Finally, one determines the mean value P_{grel} of the total pressure in the relative system for the single axis distance by integration of values measured over the blade section by using the process of mass approximation of a mean, as described by G. Hubert [13] e.g. The P_{grel} values so obtained are reduced to the absolute system in the following way:

For an incompressible fluid, the following is true in the relative system[4] (Relative Index)

$$p_{grel} = p + \frac{\varrho}{2} w^2 \qquad (2)$$

and in the absolute system (Absolute Index)

$$p_{gabs} = p + \frac{\varrho}{2} c^2 \qquad (3)$$

[4]The asterisks used in equation (1) were omitted, since the values here are based on another state of rest.

with p as the static pressure, q the density both with ω as the relative and c as the absolute flowrate. The static pressure p is the same in both reference systems. Further, as one easily can show, the following obtains:

$$w = \sqrt{u^2 - 2\,u\,c_u + c^2} \tag{4}$$

with u as the local peripheral velocity of the blades and c_u the peripheral component of the absolute velocity c, which results from

$$c_u = u - \sin\,(\beta - 90°)\sqrt{\frac{2}{\varrho}\,(p_{grel} - p)} \tag{5}$$

with β the angle between ω and u. From Equation (2) to (4) it follows:

$$p_{gabs} = p_{grel} + \frac{\varrho}{2}\,u^2\left(2\,\frac{c_u}{u} - 1\right) \tag{6}$$

From Equations (5) and (6), the total pressure in the absolute system can be calculated.

From the flow angles measured over a blade section, a mean value could also be derived, but this was not done. Within the blade-wake wave troughs, in which a shear flow prevails, the pressure distribution around the cylindrical probe is unsymmetrical with regard to the direction of flow. If the angle between the two measuring bores of the probe is 90°, as in the case treated here, then considerable error may arise in the measurement of the flow angle (cf. [11]). Moreover, radial secondary currents of unknown dimension are contained in the wake wave troughs, which may cause additional falsification of the measurements. Therefore, to determine the averaged relative flow angles, the values measured outside of the troughs are used exclusively. This procedure is based on the assumption that the actual angle within the wave troughs do not change, or that deviations from a constant value are cancelled out in the mean.

In comparing the angles measured with stationary and rotating probes, the procedure was reversed from that for total pressure: here the values obtained in the absolute system were recalculated into the relative. Here the relationships derived directly from the velocity triangle are used.

5. Discussion of the Results

5.1 Results of the Wake Measurements

For the three relative axial distances r/r_A = 0.590, 0.703 and 0.950, Figures 7a to 7d through 9a to 9d show the total pressure distributions obtained with the rotating probe over one blade section at the four rpms. These were calculated from the measured values according to Equation (1) and were based on the kinetic pressure $\varrho^* u_A/2$ of the peripheral velocity u_A at the blade tips. Figures 10a to 10u show the angular distributions measured at all five axial distances.

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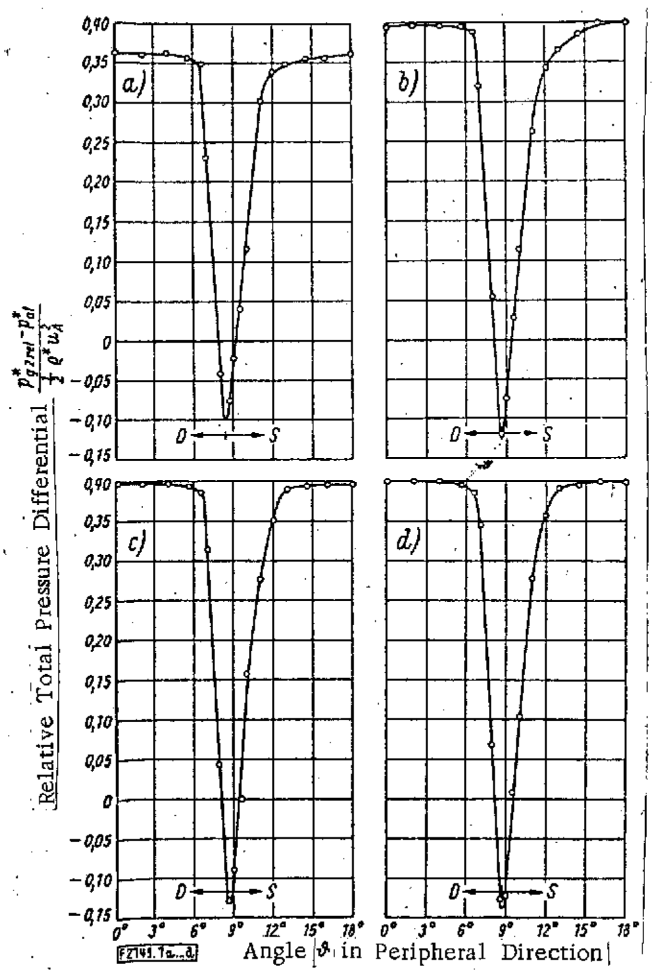


Figure 7a - 7d. Dimensionless Total Pressure Distribution in the Relative Axial Distance r/r_A = 0.590.

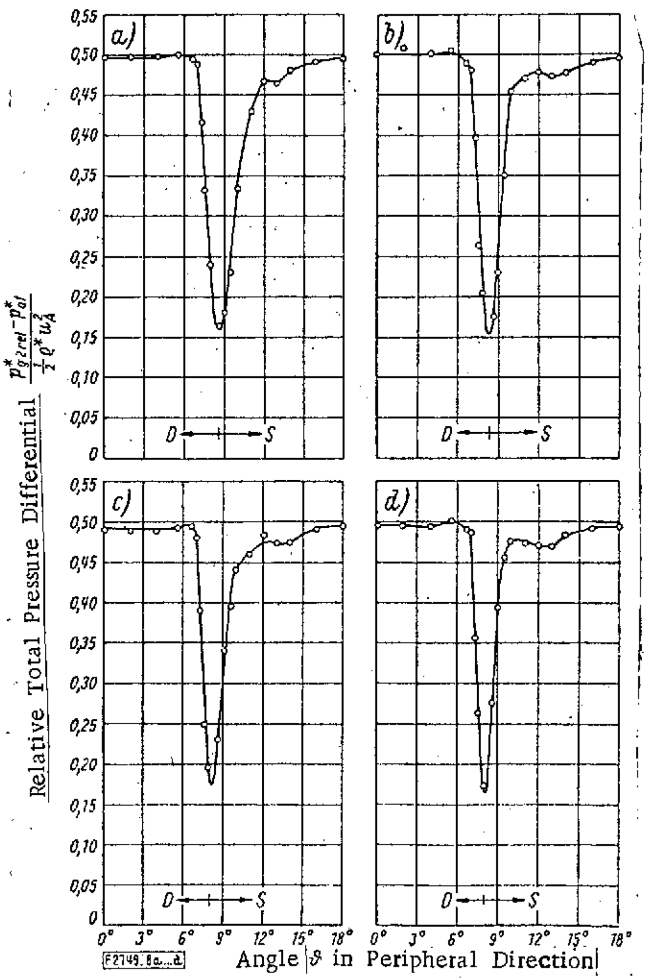


Figure 8a - 8d. Dimensionless Total Pressure Distribution in the Relative Axial Distance r/r_A = 0.703.

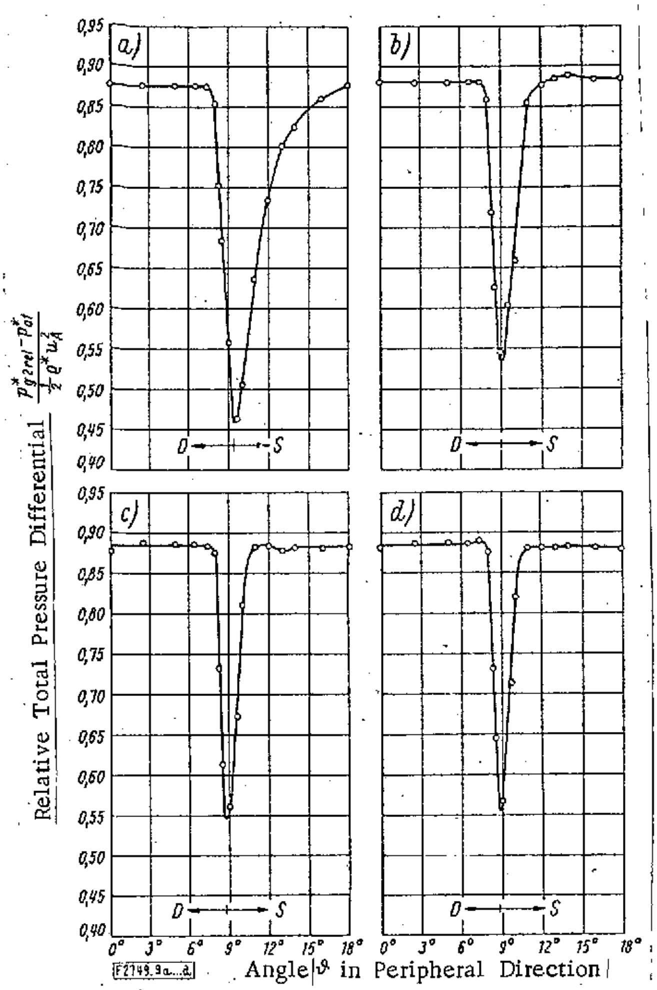


Figure 9a - 9d. Dimensionless Total Pressure Distribution in the Relative Axial Distance $r/r_A = 0.950$.

Figure 7a to 7d through 9a to 9d. Distributions of the Relative Total Pressure Over the Blade Section Measured by the Rotating Probe. p^*_{g2rel} and p^*_{at} Total pressure behind the rotor in the relative system and ambient pressure; $q^*u_A^2/2$ the kinetic pressure formed with the fluid density q^* and the peripheral velocity u_A of the blade tips (* signifies uncorrected stationary condition), ϑ angle in peripheral direction, D pressure side, S suction side; a) rpm n = 1000; b) n = 1350; c) n = 1700; d) n = 2050

In comparing wake wavetroughs, /31 their volume in the section nearest the hub (Figures 7a - 7d and 10a - 10d) at $r/r_A = 0.590$ is most noticeable. These relatively high losses are attributed to the fact that the blades already lie within the hub edge layer. One may also discern immediately a marked effect of the rpm, which obviously connected to the Reynolds number. It becomes especially evident in the section next to the outer housing at $r/r_A = 0.950$. There, for n = 1000 rpm, the wavetrough is very large and considerably expanded on the suction side, Figure 9a. Although raising the rpm to 1350 rpm causes only a relatively small increase in the Reynolds number, the losses drop by about half, Figure 9b. No more large changes result at further rises in rpm. A similar phenomenon is recognized in the section at $r/r_A = 0.590$. At n = 1000 rpm the total pressure no longer reaches the same relative maximum as for the higher rpm; at n = 1350 rpm there is still a considerable widening of the trough on the suction side. The two tendencies are noticeable in the process shown in Figure 11 for the local relative mean of the total pressure versus the relative axial distance[5].

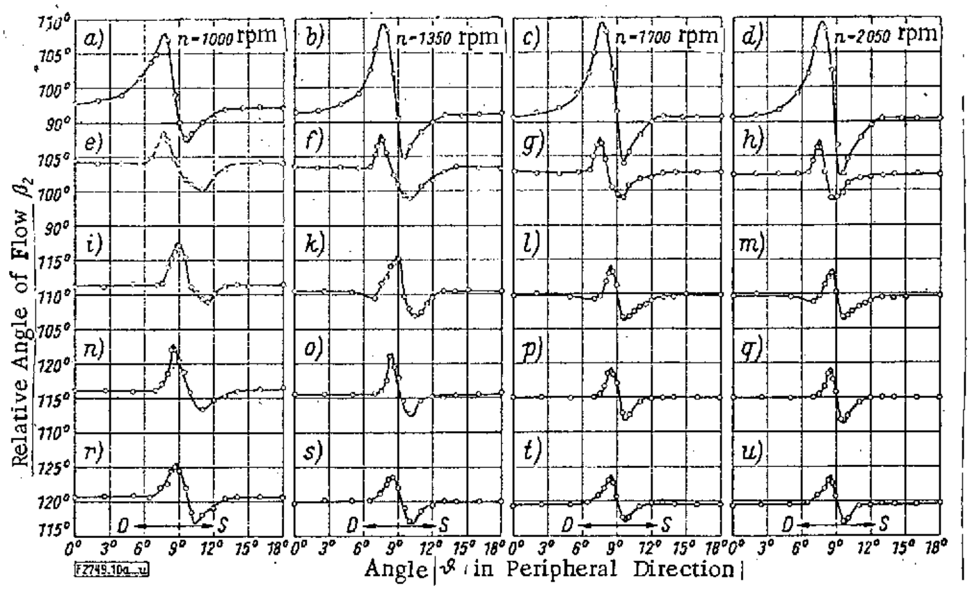
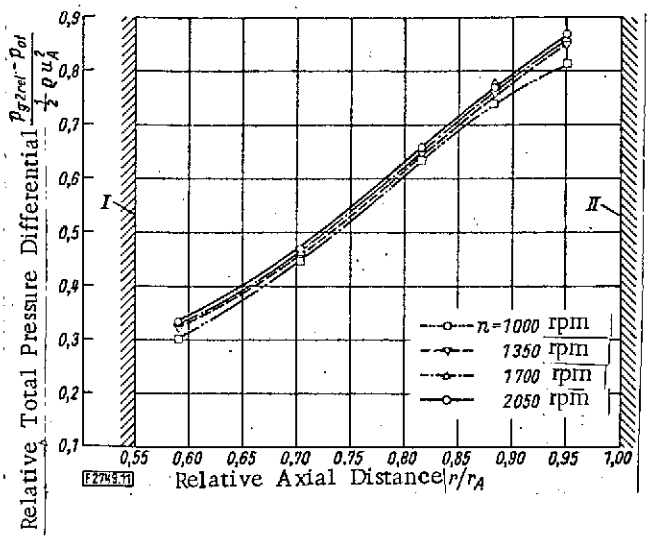


Figure 10a - 10u. Distribution of the Flow Angle Over the Blade Section Measured With the Rotating Probe. β_2 Angle between the relative flowrate behind the rotor and the blade peripheral velocity; a - d) for the four rpm's n = 1000, 1350, 1700 and 2050 rpm at relative axial distance r/r_A = 0.590 (with r as axial distance and r_A as radius at the blade tips) e - h) for the same four rpm's and r/r_A = 0.703; i - m) for the same four rpm's and r/r_A = 0.815; n - q) and r - u) the same but for r/r_A = 0.815, 0.885 and 0.950.

In Figure 12 the curve of the relative flow angle, measured with the rotating probe, is plotted for given axial distances. Here too, the combined rpm and Reynolds number influence is shown very clearly: between 1000 and 2050 rpm the displacement in the rotor near the hub increases by about 1.6° and near the outer housing by another 0.7°. Plotting the absolute flow angles α_2 determined with the fixed probe and the total pressure p_{g2abs} in the absolute system (behind the

[5]These values already have been calculated for the resting state, which prevailed in tests with the fixed probe; the asterisk is thereby eliminated.

rotor) over the given axial distance in Figures 13a and 13 b yields similar assertions.



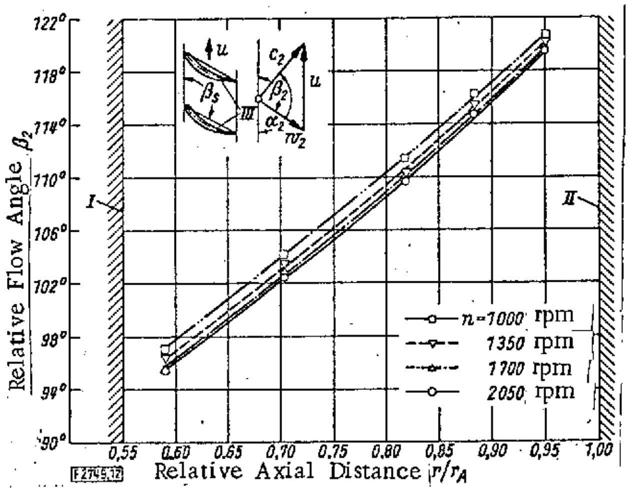
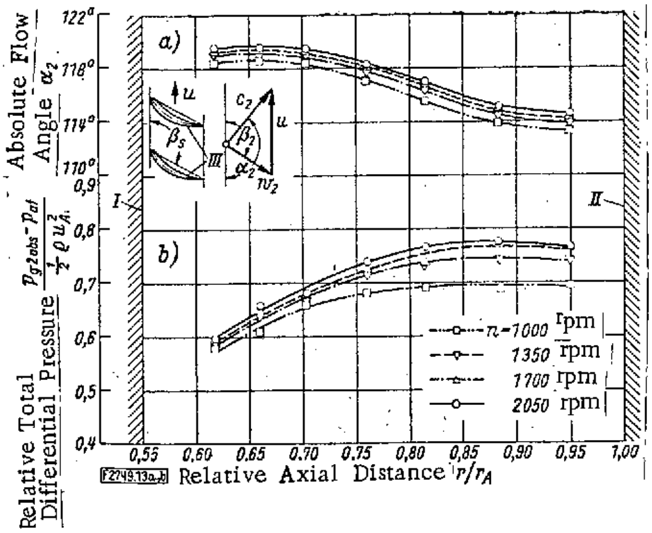


Figure 11. Distribution of the Relative Total Pressure.

Figure 12. Distribution of the Relative Flow Angle.

Data for Figures 11 and 12 Measured with Rotating Probe. p_{g2rel} and p_{at} are total pressure behind the rotor and peripheral pressure; $qu_A^2/2$ is kinetic pressure formed at the blade tips, with q the fluid density at peripheral velocity u_A; β_2 is angle between the relative flowrate ω_2 behind the rotor and its peripheral velocity u; β_s is blade angle; r is rotor axial distance, r_A is radius at blade apex; n is rpm; I is hub; II is housing; III is blades.

The results plainly illustrate part of the difficulty extant in converting data obtained in a grid wind tunnel to a rotating blade. Near the hub, the Reynolds numbers at the lowest rpm are so low that the blade boundary layer would certainly dissolve with a stationary grid in the wind tunnel. With the rotating rotor, in contrast, this does not occur. This positive influence of the blade rotation can be attributed to the centrifugal and Coriolis forces, which above all have an effect on the inner layers lying near the hub similar to a boundary layer suction. Many authors have described this effect, e.g. [4, 14-16]. /32
A. Betz [17] has tried to generalize from the results of H. Himmelskamp [15].



Figures 13a and 13b. Distributions of the Absolute Flow Angles Measured with the Stationary Probe and the Relative Total Pressure Over the Pertinent Axial Distance. p_{g2abs} Total pressure behind the rotor in the absolute system; other legends as in Figures 11 and 12; distributions of the: a) absolute flow angle, b) total pressure.

5.2 Comparison of Results Measured with Rotating and Stationary Probes

Figures 14 and 15 contain the comparisons for total pressure and flow angle measured with rotating and stationary probes. In addition, the measurements have been summarized in Table 1 for easy visibility.

The total pressures agree remarkably well. Greater differences, aside from a deviation of 3.7% for r/r_A = 0.590 and n = 1000 rpm, appear only in the section where r/r_A = 0.703. These are approximately 7%, 5%, and 3% for n = 1000, 1350 and 1700 rpm. In this regard, it should be noted that the accuracy of measurement, in general, decreases with falling rpm. On the one hand, the absolute reading at the manometer is then lower; on the other hand the water column fluctuates considerably over time in the tests with rotating probe. It is true that temporal averages were obtained, yet especially large errors naturally attended this because of the smaller absolute values at lower rpm. On the same grounds, the accuracy of results with the rotating probe in comparison with the fixed probe decreased toward the hub. Although the difference between the mean total pressure and the ambient pressure near the housing in the relative system has the same order of magnitude as in the absolute, this difference at the hub was only about half as great and was very small in the wake troughs. The deviations in Figure 14 apparently are the result exclusively of measurement inaccuracies, and not from a pulsating stream over the fixed probe.

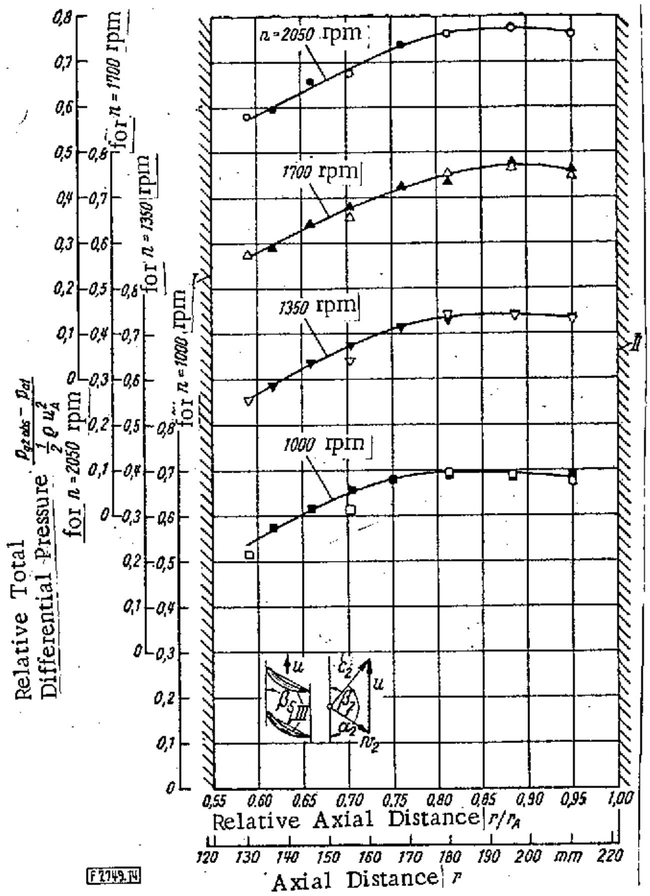


Figure 14. Distributions of the Absolute Pertinent Total Pressure.

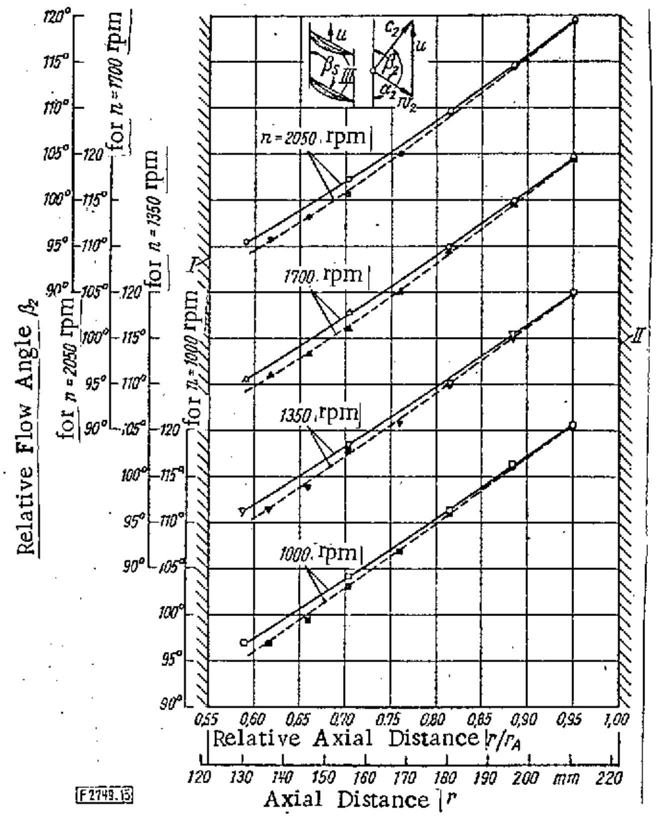


Figure 15. Distributions of the Relative Flow Angle.

Figures 14 and 15. Comparisons of the distributions obtained with the rotating and the fixed probes of the pertinent total pressures and relative flow angles over the pertinent axial distance.

Legends as in Figures 11, 12, 13a and 13b; open symbols for a corotating probe, closed symbols for a fixed probe.

There is a considerably different outcome in comparing the relative flow angles determined by the two probe arrangements, Figure 15. Although agreement is very good in the outer sections measured, the deviations increase toward the hub in an apparently systematic way. In the section nearest the hub they are greatest at all rpm, equalling up to 2°. /33

TABLE 1. COMPARISON OF THE PERTINENT ABSOLUTE TOTAL PRESSURES AND RELATIVE FLOW ANGLES DETERMINED WITH THE ROTATING AND FIXED PROBES.

Symbols as in Figures 11, 12, and 13a and b.

Relative Axial Distance $\frac{r}{r_A}$	Revolutions n rpm	Relative Flow Angle β_2		Total Pressuse Difference $\frac{p_{g2abs} - p_{at}}{\varrho u_A^2/2}$	
		Rotating Probe	Space-bound Probe	Rotating Probe	Space-Bound Probe
Rotating probe: 0,590 Fixed probe: 0,617	1 000 1 350 1 700 2 050	97,0° 96,3° 95,6° 95,4°	96,9° 96,4° 95,9° 95,7°	0,517 0,556 0,574 0,582	0,579 0,587 0,590 0,596
Both probes: 0,703	1 000 1 350 1 700 2 050	104,2° 103,5° 102,8° 102,5°	103,1° 102,9° 100,9° 100,6°	0,613 0,642 0,656 0,676	0,660 0,677 0,680 0,681
Both probes: 0,815	1 000 1 350 1 700 2 050	111,3° 110,3° 109,9° 109,7°	111,1° 109,9° 109,4° 109,4°	0,697 0,745 0,753 0,765	0,690 0,732 0,735 0,765
Both probes: 0,883	1 000 1 350 1 700 2 050	116,3° 115,5° 114,9° 114,7°	116,1° 115,1° 114,3° 114,3°	0,692 0,741 0,767 0,774	0,683 0,742 0,771 0,777
Both probes: 0,950	1 000 1 350 1 700 2 050	120,7° 120,0° 119,7° 119,5°	120,4° 119,9° 119,4° 119,4°	0,673 0,735 0,746 0,761	0,696 0,740 0,764 0,765

Commas indicate decimal points.

W. M. Schulze et al [4] had already obtained similar results. In their studies they also found no deviation between the angle reading of rotating and a fixed probe toward the outside behind a compressor rotor but to the inside there was a difference of about 1°. While W. M. Schulze et al [4] limited their measurements to one rpm reading, the results here show that these differences do not depend on the rpm and therefore are not dependent on the frequency of the pressure perturbations in front of the fixed probe. From this it would be proper to conclude that a false formation of the temporal mean cannot be the cause of the differences in Figure 15. Even changes in amplitude and form of the perturbation appear to play no role. Therefore are considerably different in their amplitude and form at the blade tips at the smallest axial distance used, at which one obtains the greatest discrepancies in angles; but at other axial distances their differences are so small that one cannot thereby explain very well the systematic increase of the angle deviations from outside to inside. It also seems improbable that the relatively short measuring bores of the probe in the comparison to the total pressure measurement could be responsible for this.

On the other hand, these angle deviations can scarcely be referred to as pure measurement inaccuracies. Granted that the radial total pressure gradient could have a certain influence in this respect. For one thing, the influence in the relative reference system is considerably greater than in the absolute, and for another a probe set in the hub has the opposite direction from that mounted in the housing. According to studies of E. Becker [18] the maximum slope of the static pressure distribution at the periphery of a cylindrical probe of the design used here depends considerably on the shearing flows in the direction of the probe axis and, in fact, both on the magnitude of the velocity gradients and on their direction (compare [11]). This is due to the fact that secondary flows occur, which change the dead water pressure behind the probe. This does not necessarily cause errors in angle measurement, but their accuracy is much influenced. Comparison of the results of E. Becker [18] with the total pressure distributions in Figures 11 and 13b shows that the sensitivity of angle measurement with the rotating probe over the whole range of axial distances is lower than with the fixed probe, with which it actually increases toward the hub. In addition, the relative velocities at small axial distances are considerably smaller than the absolute velocities, so that even the pressure differences at the two measuring bores of the rotating probe, with equal deviation of the flow direction from the symmetry axis, assume lower values than at the bores of the fixed probe. Thus it could be expected that the measured points obtained with the rotating probe would diverge more with decreasing axial distance, because of the reduced accuracy. The systematic form of the deviations in Figure 15, however, are difficult to explain by this.

However, there is still the question of whether the procedure chosen here to compare the angles obtained from the two different methods is adequately justified. As was described, it was assumed that the angles within the wake troughs do not change, and only the angles were used which the rotating probe indicated outside the troughs. But if real changes should occur in the angle β_2 in the wakes of the blades in compressors, then this procedure would be fraught with error. There are no experimental results whatever known to the author which answer this question. A final solution of the problem therefore could only be obtained by carefully measuring the wake behind a blade-grid with a directional probe resistant to shear flows. According to K. Wieghardt [19] a cylindrical probe with its angle bores displaced some 60° from each other would fulfill this requirement, if a perfect theoretical pressure distribution existed at its circumference. In the actual, friction-affected flow this angle would probably be greater (cf. [11]) and have to be determined experimentally.

After one had developed such a probe, measurements could be made relatively simply with it behind a stationary blade-grid. With a rotating blade-wheel the radial secondary flows within the wake troughs would cause considerable difficulty. One then would first have to measure the additional oblique flows in the rotating system which the secondary flows cause on the probe to be able to suitably correct the angle reading. So long as such fundamental studies are not available, the problem of possible error in the angle measurements with pulsating flows over the probe can hardly be cleared up.

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